DYNASHAM TRUCK COMPANY, LTD.* (A)

Truck Suspension Bolt and Bushing Failures

In March 1958 Lance O'Leary, a Project Engineer at the Dynasham Truck Company, Ltd., Chicago, Illinois, was considering possible solutions to the problem of persistent fractures of a truck suspension pivot bolt and failures of its rubber bushing. The failures occurred on heavy duty trucks equipped with Dynasham's air suspension system.

Dynasham was established in 1922. The company builds many different models of heavy duty trucks for both highway and off-the-road use. Engines (almost all diesel), transmissions, other drive train components and many other parts are purchased. However, Dynasham's trucks are often custom engineered to suit customer requirements. Many parts are designed and built by Dynasham or their suppliers as needed. In the interests of light weight, aluminum is used for body panels, in wheels, bumpers, radiators, mounting brackets for engines, transmissions and other components, and even for the frames of some models. The company employs about 2500 people; in 1965 they produced more than 10,000 trucks.

*All dates and names of people, places, and firms in this case study have been disguised, except as noted.

Prepared in the Design Division, Department of Mechanical Engineering, Stanford University, by John A. Alic under the direction of Professor H. O. Fuchs, with support from the National Science Foundation.

take vertical loads, lateral loads, and braking and accelerating loads. Lance said, "The arms are springs rather than rigid links because springs will bend under heavy shock loads -- they give an additional safeguard for the truck and its payload. Also, a spring will bend where a rigid link might break."

Lance's problem was concerned with fractures of the pivot bolts, item 17, and with deterioration of the rubber bushings, item 19, which wore out quickly. Two of these bushings fit inside each adjustment block-radius rod, item 1. This block bolts to the lower leaf of the spring. Shim washers, item 16, are used to adjust its position with respect to the spring hanger, item 13, which is bolted to the frame. A section view of the bushed connection is shown in Exhibit 3. Details of the spring, adjustment block, and spring hanger are shown in Exhibits 4 through 6. Both the spring hanger and the adjustment block are steel castings and are used with unmachined bores. At assembly, 2/3 to 3/4 of the length of each bushing is dipped into a soap compound for lubrication, two bushings are used for ease of assembly. When forced into the bores the bushings are compressed. Inserting the 7/8 inch bolt causes rubber to be extruded out both ends of the annular space. When the nut is tightened, the washers, item 20, on the outside of each leg of the spring hanger, trap the rubber and force it back into the cavity. The assembly specification calls for the nut and bolt to be tightened until the washers are pulled tightly against the hangers, but not so tightly that the gaps between the adjustment block and the hanger legs are closed. These gaps, at either side of the block, are supposed to total about 1/16 inch. No gauging was being done at assembly.

The rubber bushings serve several purposes. The spring must be stiff so that it will not flex inordinately when subjected to accelerating or braking torques. The bushings are therefore called upon to absorb some of the road shock and to isolate the spring from the frame. They permit the two castings to be used with unmachined bores and they also allow a small amount of lateral axle motion, after which the adjustment block contacts the spring hanger legs. Lance explained that this relieves the bolt of some of the cornering loads. A rubber bushing is also desirable because it requires no lubrication or other servicing. Lance said, "The rubber is compressed to limit motion in the fore and aft direction and because compression increases its life. We assume a bushing has failed when it is worn to the point of being sloppy -- so that excessive axle movement is allowed. This condition is not dangerous although uncertain steering often results and the controllability and stability of the vehicle may be impaired. Worn out bushings also usually cause abnormal tire wear. If bushings fail before a normal vehicle overhaul the driver usually discovers it by the effect on vehicle stability or tire wear."

"Broken pivot bolts are usually discovered while one or both halves of the bolt are still held in the rubber bushings," he continued. "Broken bolts, like worn bushings, result in excessive lateral axle movement and the effect on vehicle stability is usually apparent to the driver. Bolt failures may also be discovered during routine inspections."

Empyrean Supply has told Lance that they have never had any bolt or bushing trouble on their trailer suspensions. The trailer suspension, however, does not use the cast adjustment block; the bushings fit inside a rolled eye in the spring leaf. In addition, the trailer suspension is not subjected to driving loads.

Bushing Failures

When Lance was investigating the problem of inadequate bushing life in March 1958, about 200 air suspensions were in service on Dynasham trucks. The bushing then used as made from a 1-1/2 inch diameter cylinder of extruded neoprene in which a lengthwise hole had been drilled for the bolt. Each of the two bushings within an assembly was 3-1/2 inches long before compression; the Durometer hardness specification on the neoprene was 60 to 70. Bushings with a length of 3-3/4 inches were later used in an attempt to increase bushing life by compressing the rubber more. This helped only slightly, if at all. Typical premature failure of these bushings is shown in the photographs of Exhibit 7. Lance said, "Tearing occurred because the rubber in the adjustment block twists relative to that trapped in the hanger legs and also due to displacement of the adjustment block relative to the hanger. Both motions result in deformation of only a limited local volume of rubber. The small circular gouges on the surface of the bushings were caused by the compressed rubber working into the tapped through-holes in the adjustment block. These holes are for the shimmed capscrews at the front end of the block, which didn't extend to the ends of the holes."

Lance said that he considers 100,000 miles acceptable bushing life as this is a usual major maintenance interval and also the end of the warranty period on new trucks. Dynasham recommends that suspension bushings be replaced at 100,000 miles. He mentioned that few truckers bother to report such minor failures as prematurely worn bushings to Dynasham, but that as near as he had been able to determine, the average life of the neoprene bushings was about 20,000 miles. He estimated the range in bushing life to be from about 9,000 miles to about 50,000 miles.

When Lance began looking for causes of the bushing failures he also discovered that far from being under compression, the neoprene bushings were not even completely filling the cast holes in the two parts. He measured the bore of an adjustment block at several points with the results shown in the sketch of Exhibit 8. Obviously the bushing was not being compressed, but could work in the bore and be abraded. Lance also found that the corners of the bore, on both the adjustment block and the

hanger legs, which according to the drawings were to have a 1/8 inch radius, were actually being cast with a relatively sharp corner. This sharp corner tended to tear the rubber on the circumference of the bushing.

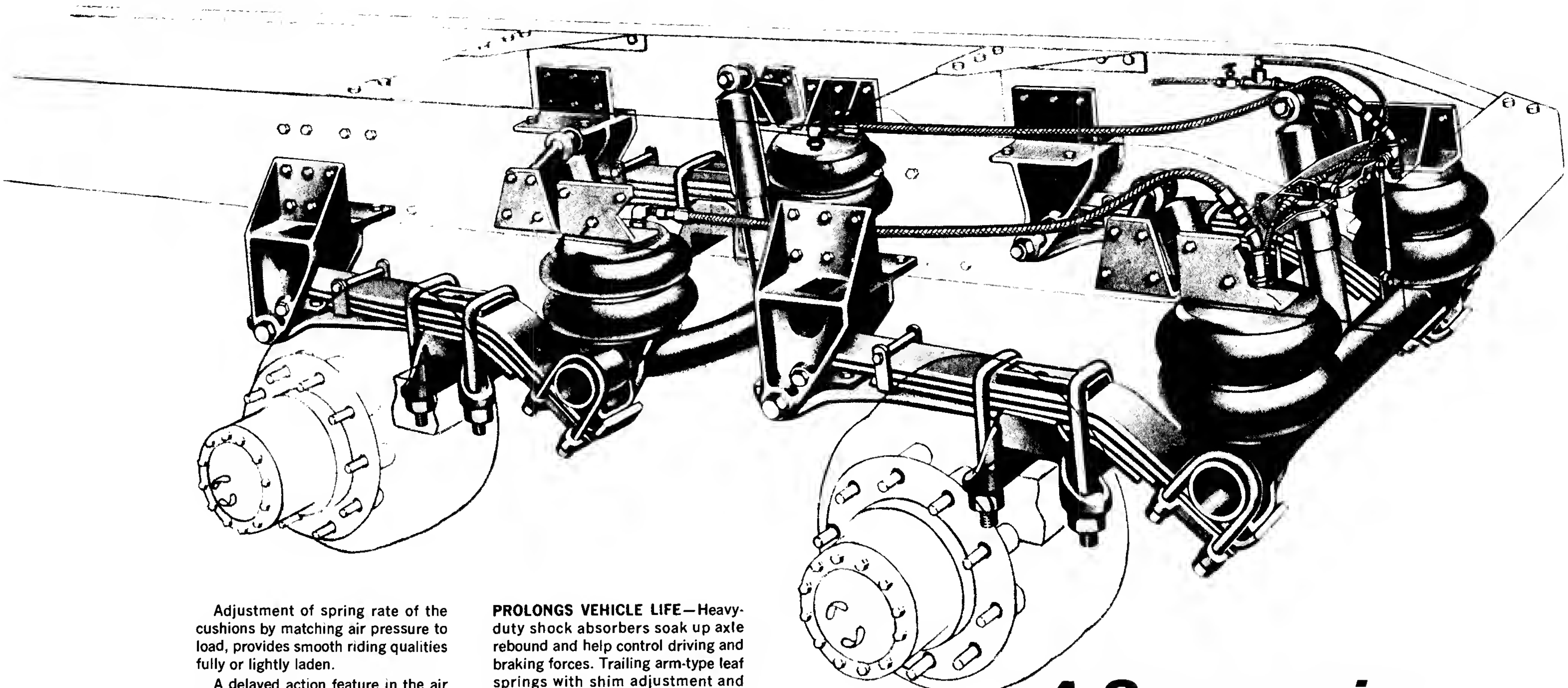
Lance knew that the cost of the new patterns which would be needed to put a cast radius on the corners of the bore of the adjustment block and spring hanger and to decrease the bore diameter would be considerable. Patterns for parts of this size usually cost $250 to $300 apiece. The steel castings themselves cost about $.50 per pound. It was felt that the bushing wear problems should be solved without going to this expense. Only if it was decided that the problems would be eliminated in no other way would the two castings be redesigned. Lance said that they wanted a satisfactory solution to the failure problems, including the bolt fractures, which would fit into the suspensions already in service with a minimum of alterations. It was desirable that parts produced in the future as replacements fit all the air suspensions previously manufactured.

Pivot Bolt Failures

Several months later, Lance attacked the problem of pivot bolt failures. Known fractures of this bolt (some no doubt were never reported to Dynasham) totaled 32. The bolts were 8 inches long beneath the head; the diameter of the shank was 7/8 inch. Threads were cut on the last 2-1/4 inches after the blank had been cold forged and its head milled to a thickness of 3/16 inch. The thin head is necessary to provide clearance for tire chains. The material specification was SAE Grade 5. Photographs of the only two fractured bolts which were returned to Dynasham and made available to Lance are shown in Exhibits 9 and 10. As far as is known, all bolts broke approximately at their midpoints. This indicated to Lance that they were failing in bending. SAE Grade 5 bolts are medium carbon steel (.28% to .55% carbon), quenched in either oil or water and tempered at 800°F minimum. They must have a minimum yield strength of 81,000 psi and a minimum tensile strength of 115,000 psi. The hardness called for in the SAE specification is Rockwell C 22 to 32 at "a transverse section through the threaded segment at midradius." When Lance ground a flat about 1/4 inch wide on the shank of one of the broken bolts and checked its hardness, he found that the hardness varied from Rockwell C 11 to RC 27 at different locations, with an average value of 20 to 21. Some of the bolts had failed after only a few thousand miles, and Lance began to suspect that a batch of inferior quality bolts might have been installed in some of the air suspensions. He proceeded to have about 70 bolts checked for hardness; only about 10% were found to be as hard as the lower limit for Grade 5.

In its original application to trailers, the pivot bolt is subject only to braking loads and to a portion of the vertical loads. Since the top three leaves of each spring bear against the arch between the legs of the spring hanger, only one leaf of the spring transfers vertical loads to the frame through the bolt and bushing. In Dynasham's application to driving axles the bolt is subject to accelerating loads, and thus reversed loading in the fore and aft direction, as well.

Lance considers acceptable bolt life also to be 100,000 miles, but Dynasham makes no recommendations to truckers for bolt replacements. When making strength computations on a truck suspension, Lance takes as the worst case for maximum load, braking with a tire/ground coefficient of friction of 0.85 or 0.90. He said that published information on numbers to use for such calculations is sparse and not very reliable. He decided upon this figure for a design load after talking with other engineers at Dynasham. He uses the allowable static axle loadings as given in Exhibit 1 (18,000 lbs. for a single drive axle and 17,000 lbs. for each axle of a dual drive set) as the normal force at the tire contact patch.



A Suspension to Smooth the Road With Any Load!

Adjustment of spring rate of the cushions by matching air pressure to load, provides smooth riding qualities fully or lightly laden.

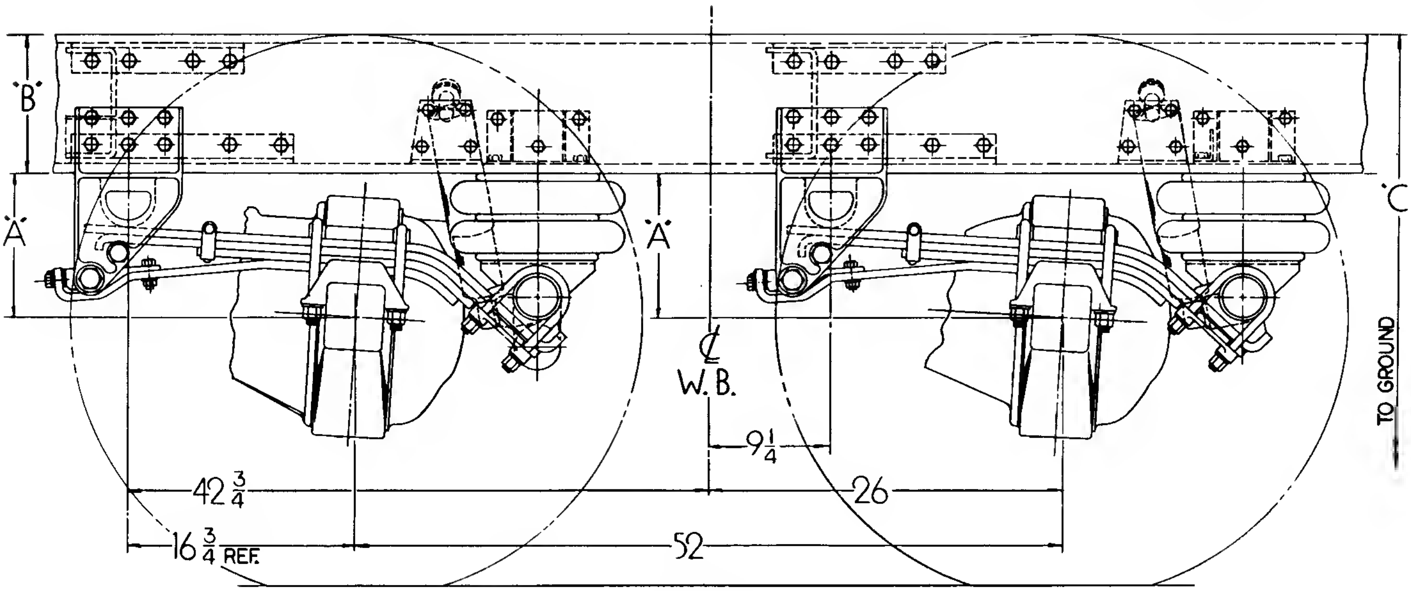
A delayed action feature in the air leveling valves prevents over correction of air pressure so that air is not wasted by minor road irregularities. The same delay feature also prevents an over correction of the load cushions on curves or when on winding roads.

On dual drive applications, a balance line between the front and rear axle load cushions distributes vehicle weight equally on each axle assuring increased tire, axle and brake lining life.

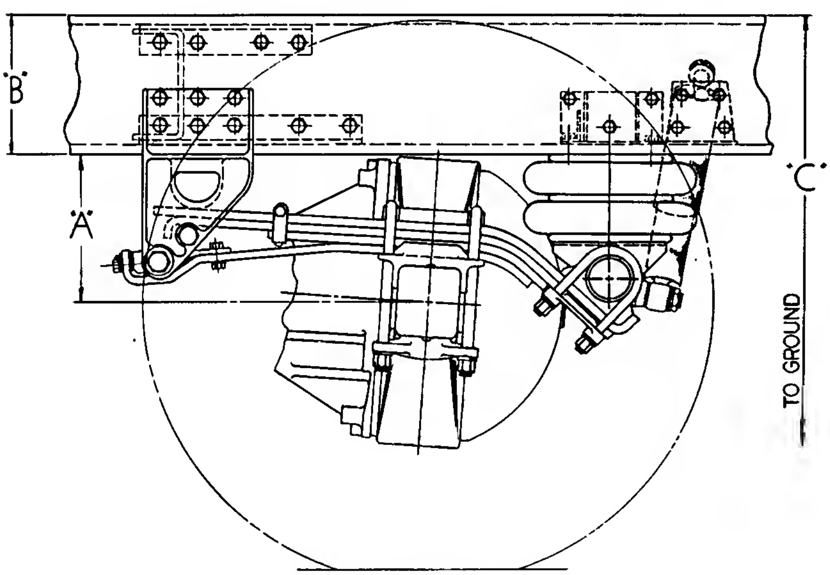
PROLONGS VEHICLE LIFE—Heavy-duty shock absorbers soak up axle rebound and help control driving and braking forces. Trailing arm-type leaf springs with shim adjustment and rubber bushed pins act as radius rods to insure positive axle alignment and resist side sway.

The 4-point suspension mounting on dual drive applications distributes frame loading over a wide area, eliminating high stress points.

Exhibit 1: The Dynasham Air Suspension System.



← FRONT



← FRONT

SQHD TANDEM AXLES			
TIRE SIZE	* DIM A	DIM B	* DIM C
11 X 22.5	10 3/4	9	40 3/8
10.00 X 20		10 1/2	41 7/8
11 X 24.5	10 3/4	9	41 1/8
10.00 X 22		10 1/2	42 5/8
11.00 X 22	10 3/4	9	41 3/8
		10 1/2	42 7/8

* ADD 1/4" FOR R-170 SINGLE AXLE

Single Drive	Dual Drive
470#	895#
18,000#	34,000#

FEATURES

Lightweight
Smooth level ride
Longer vehicle life
Excellent stability
Positive load equalization
Constant frame height
Longer tire mileage
Minimum maintenance

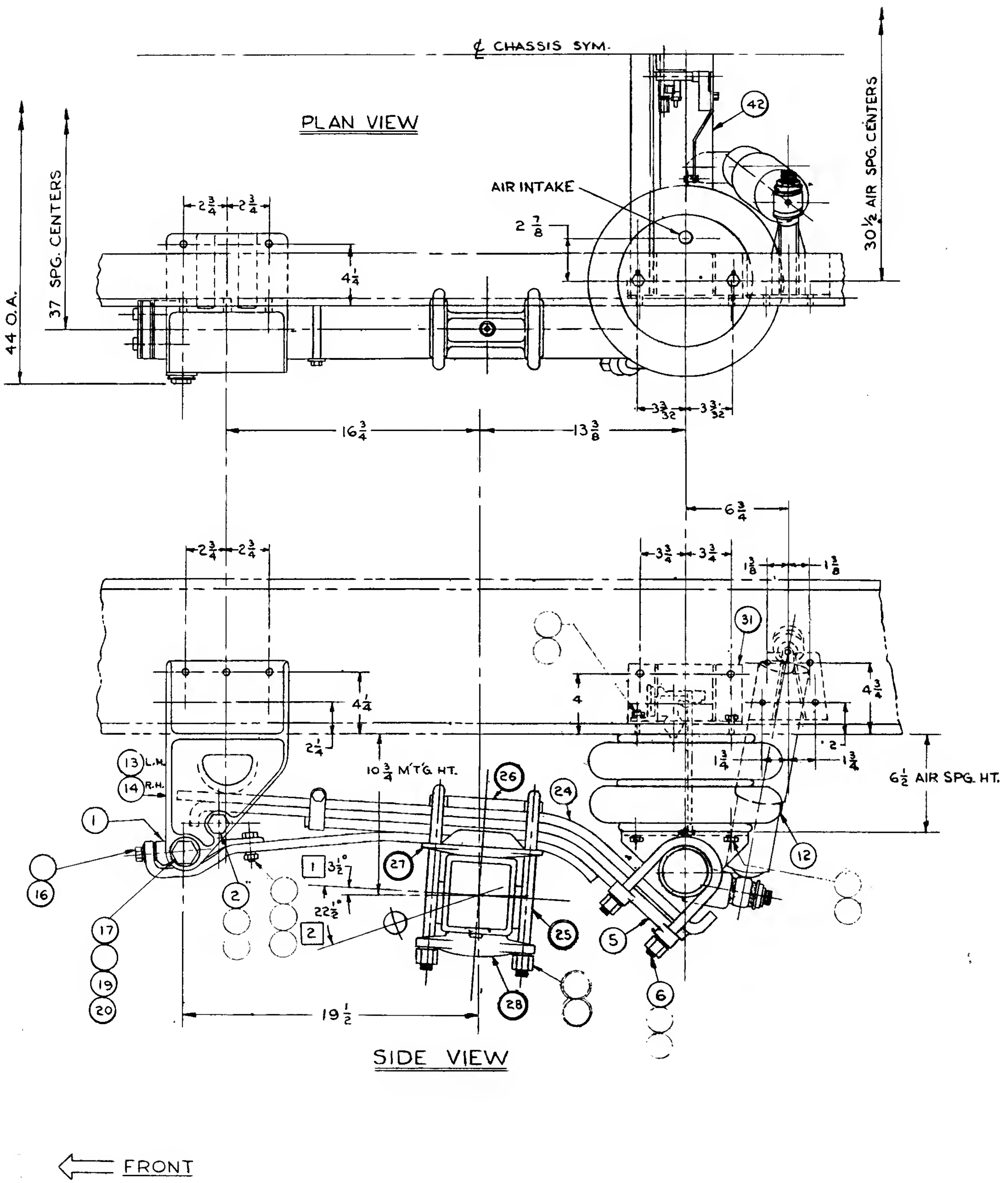


Exhibit 2: Air Suspension Assembly Drawing.

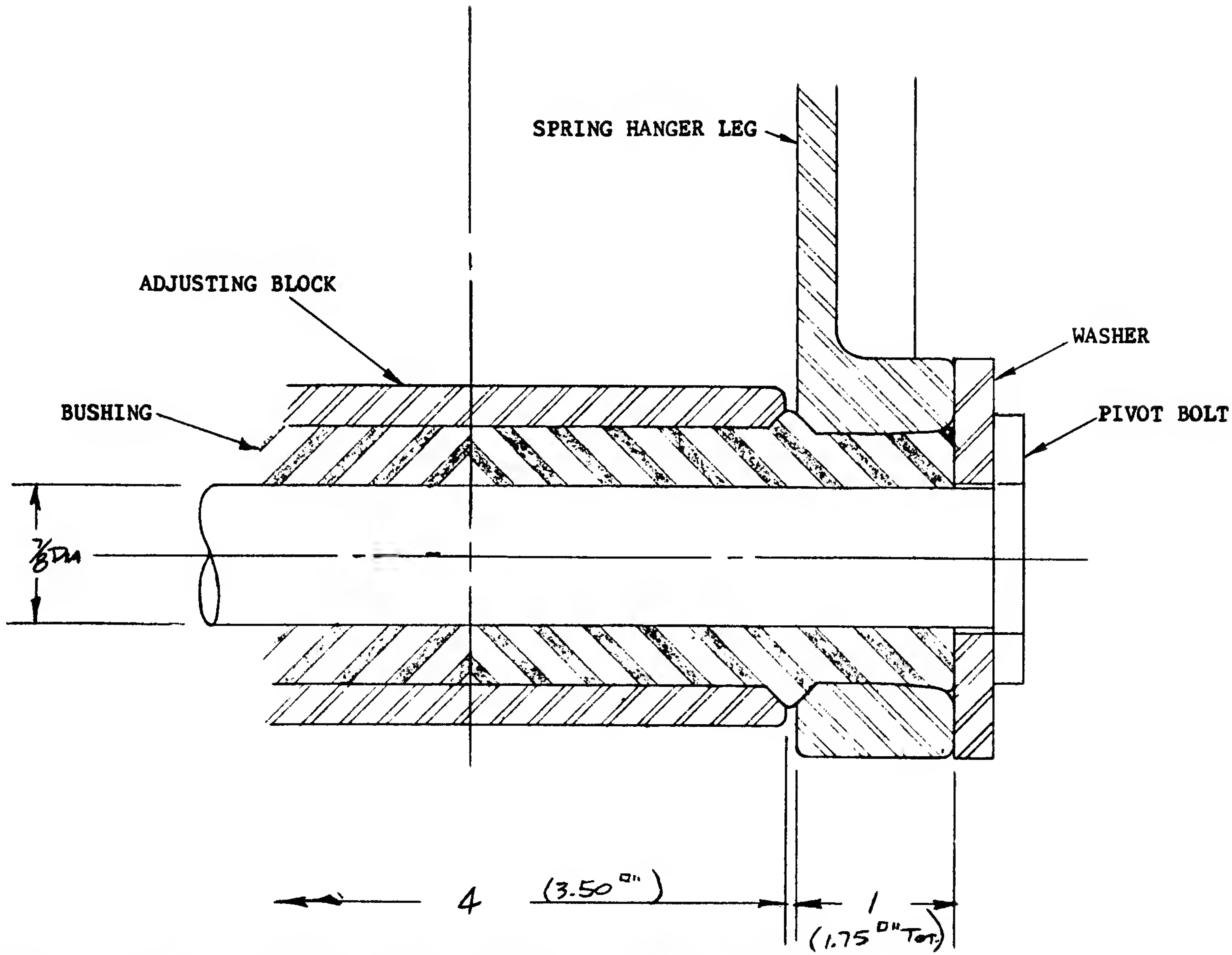


Exhibit 3: Section View of Bushed Connection. Drawn by Lance O'Leary.

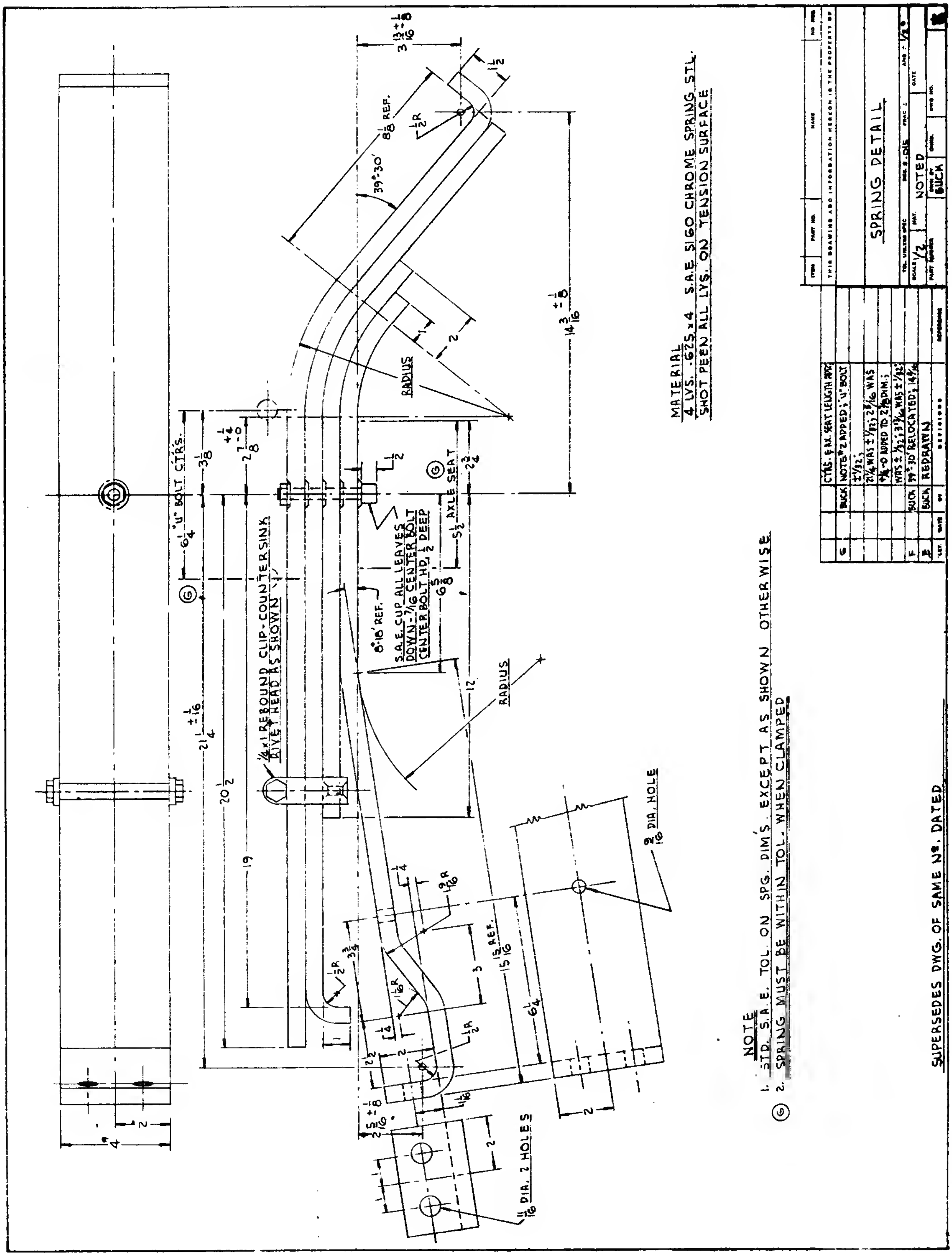


Exhibit 4: Spring.

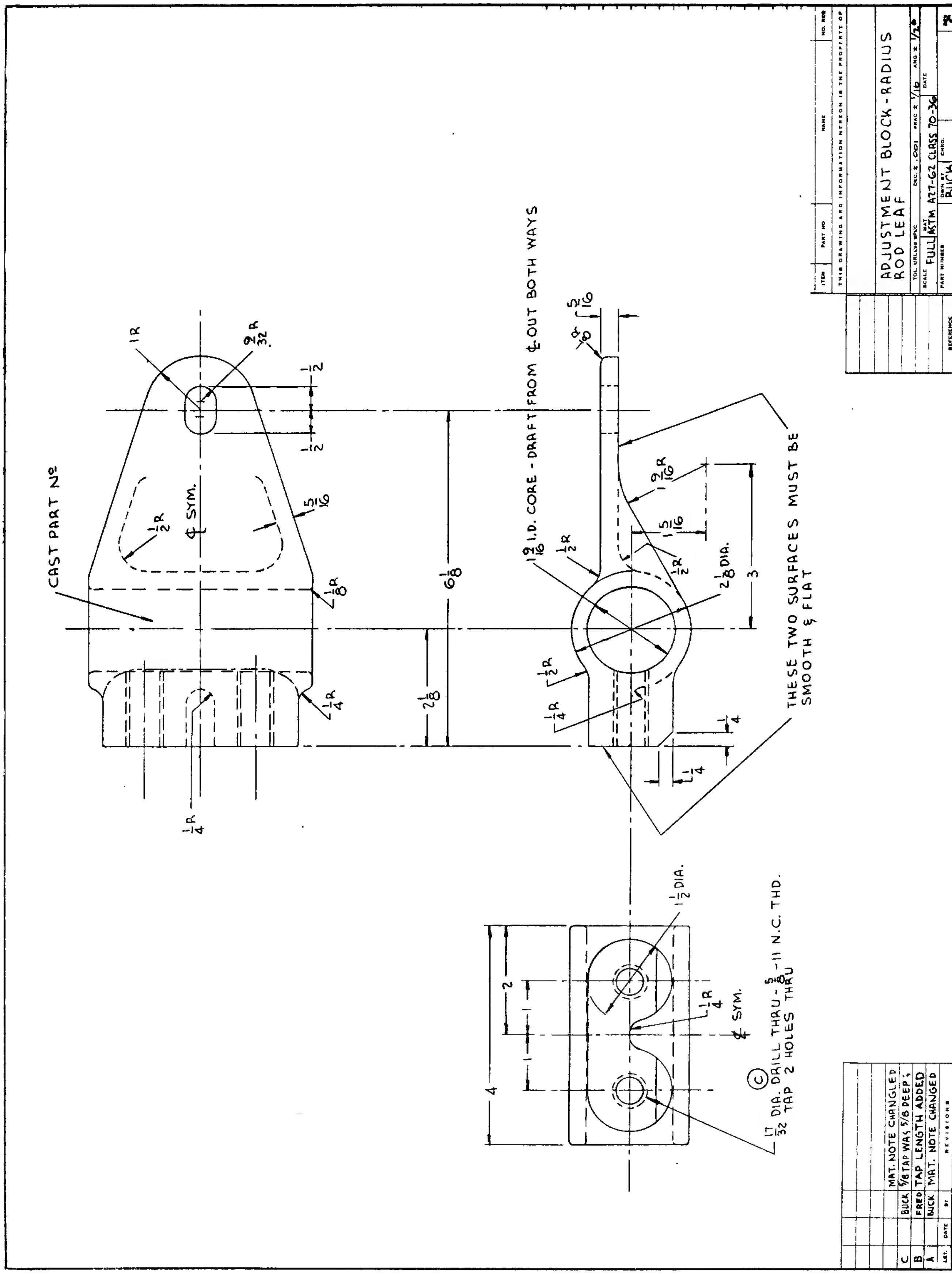


Exhibit 5: Adjustment Block.

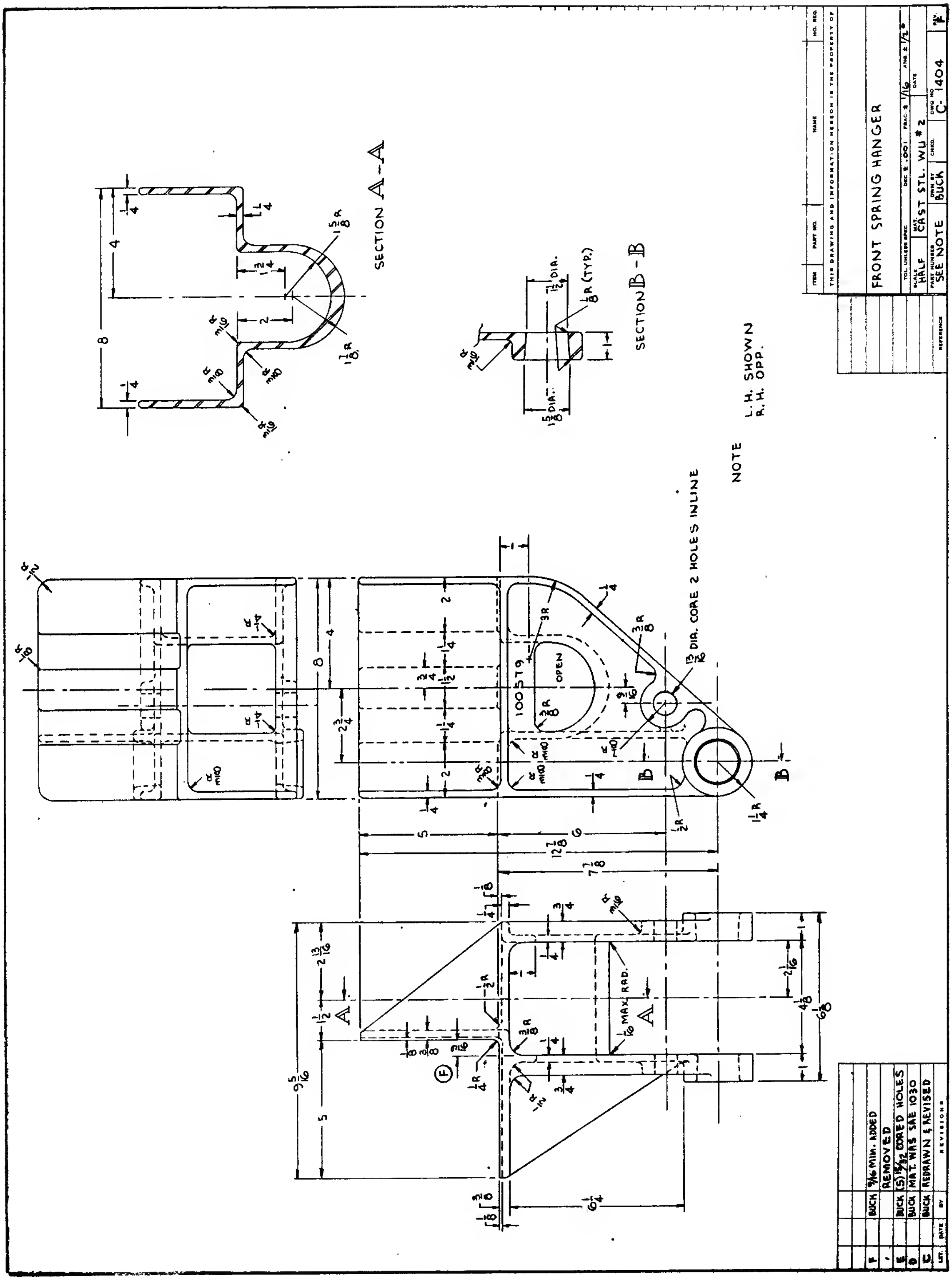
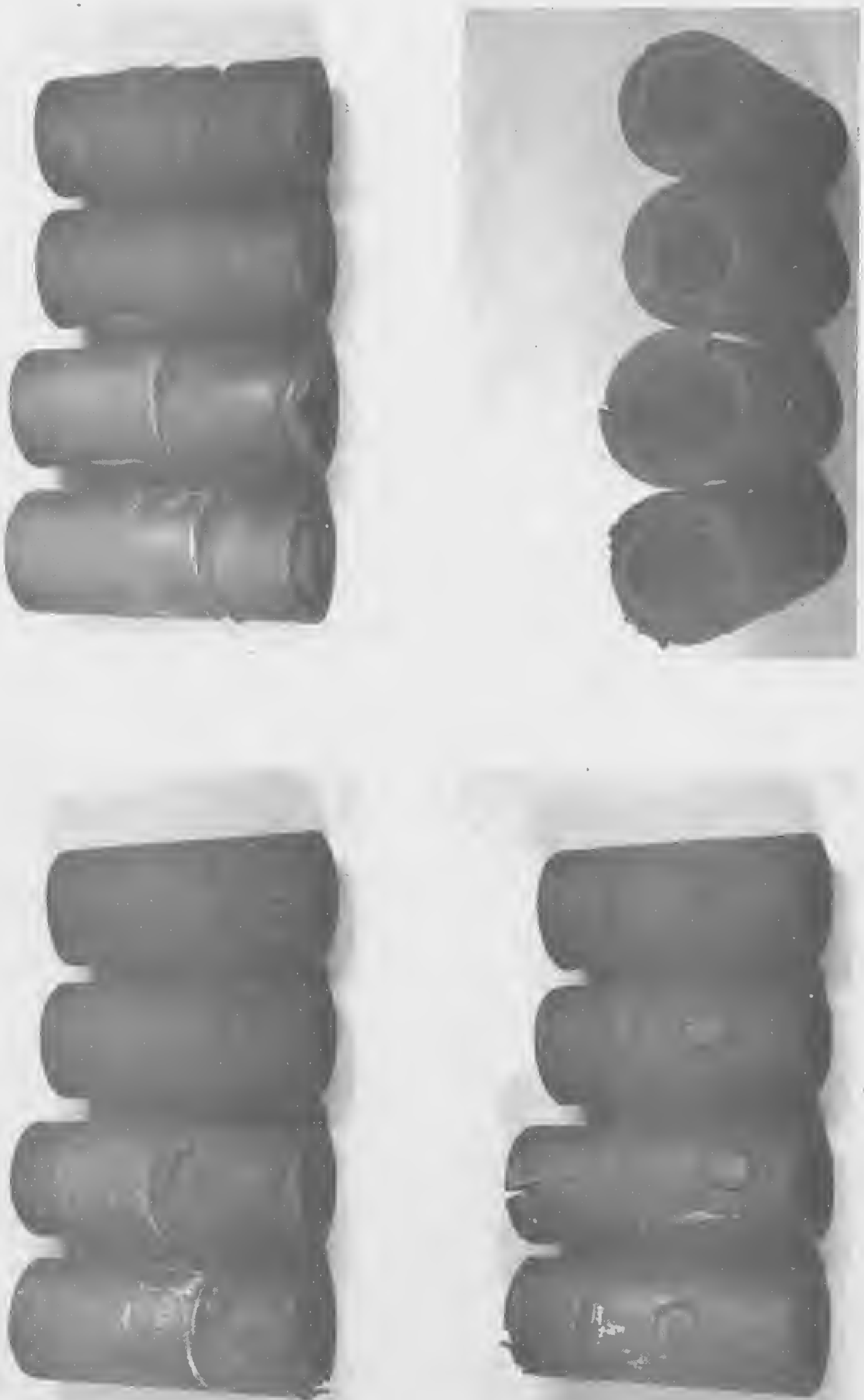


Exhibit 6: Spring Hanger.



<u>Exhibit 7:</u> Failed Neoprene Bushings. 3 1/2 and 3 3/4 inch -engths.

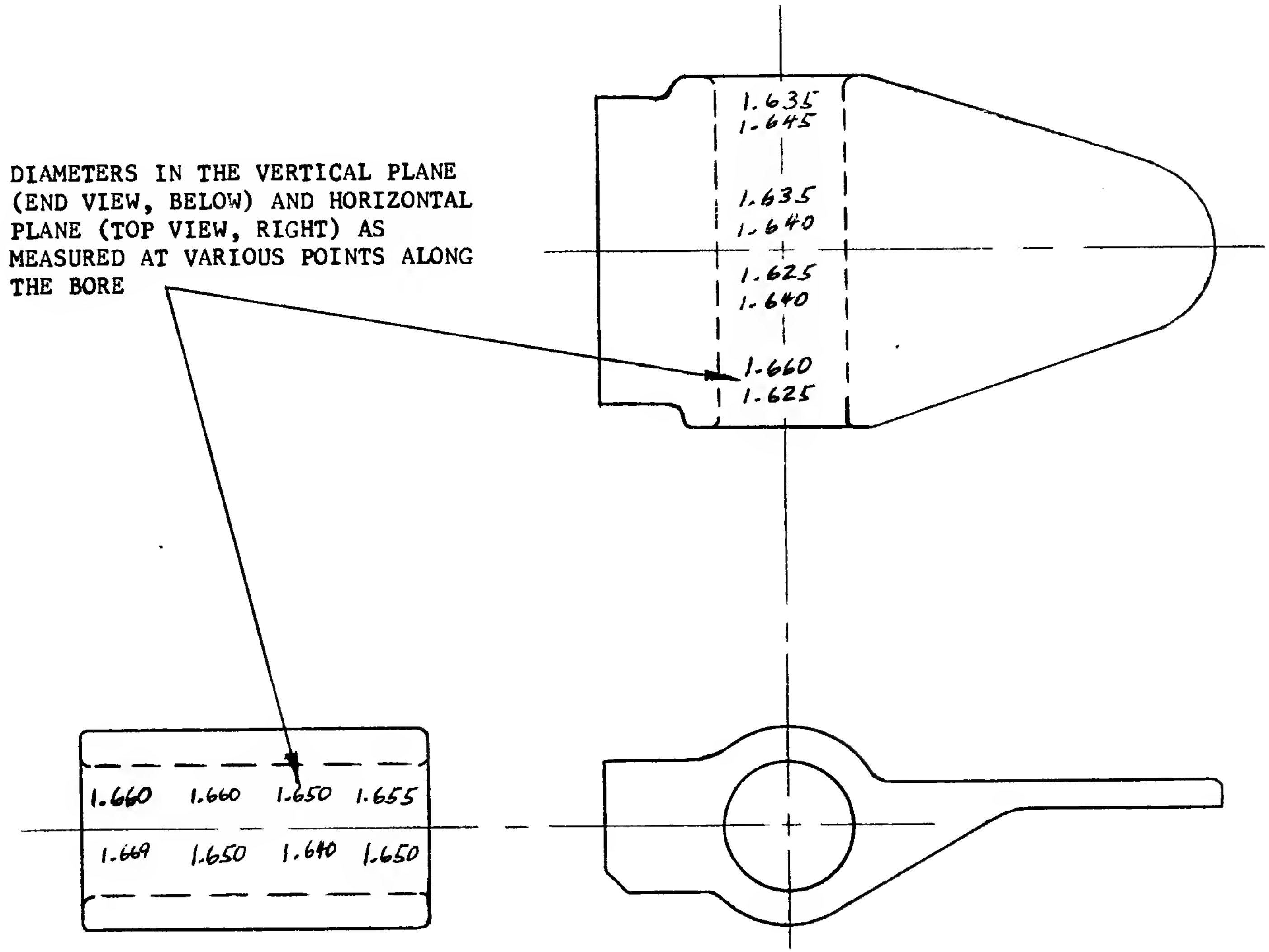


Exhibit 8: Bore Measurements of an Adjusting Block.

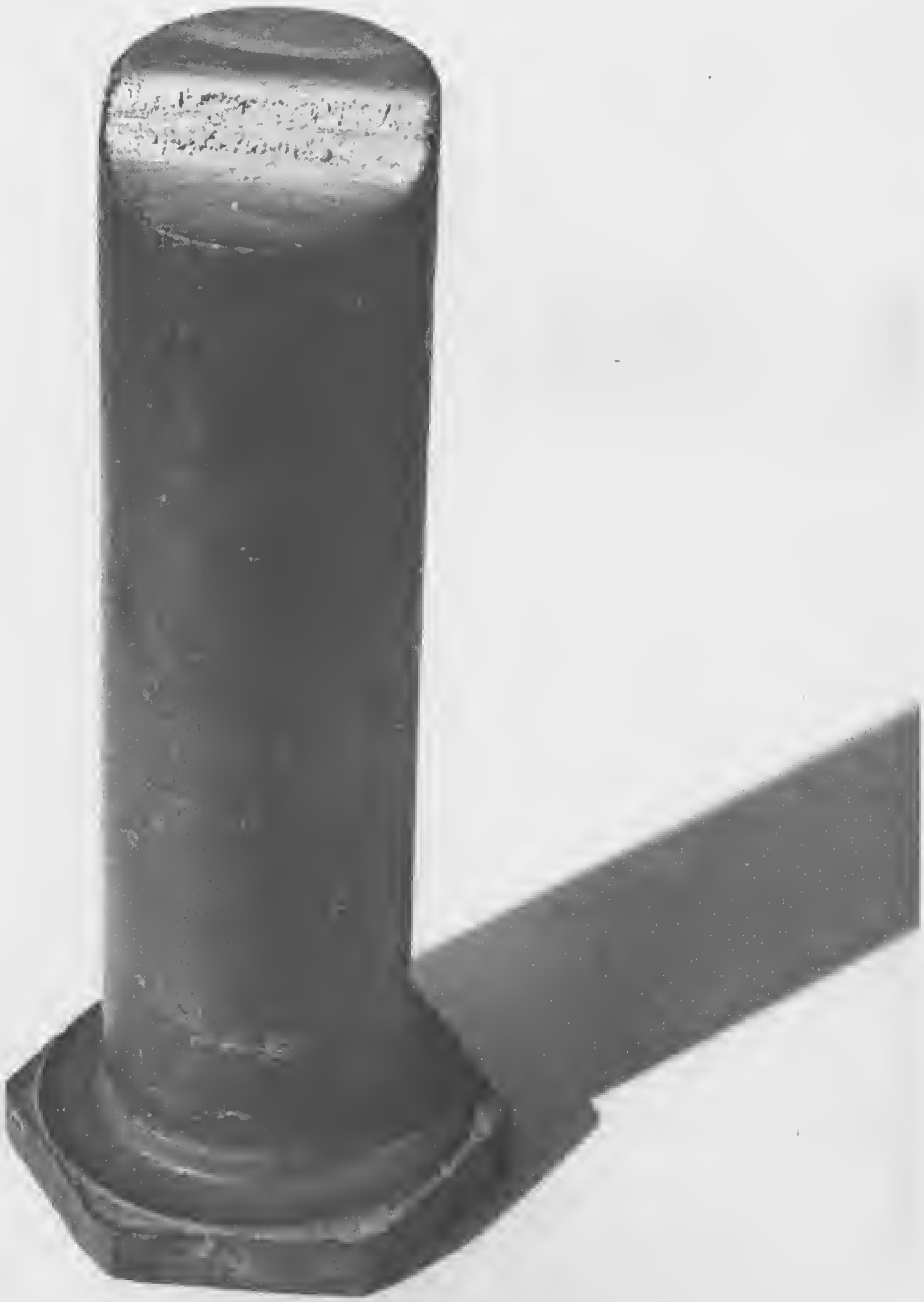
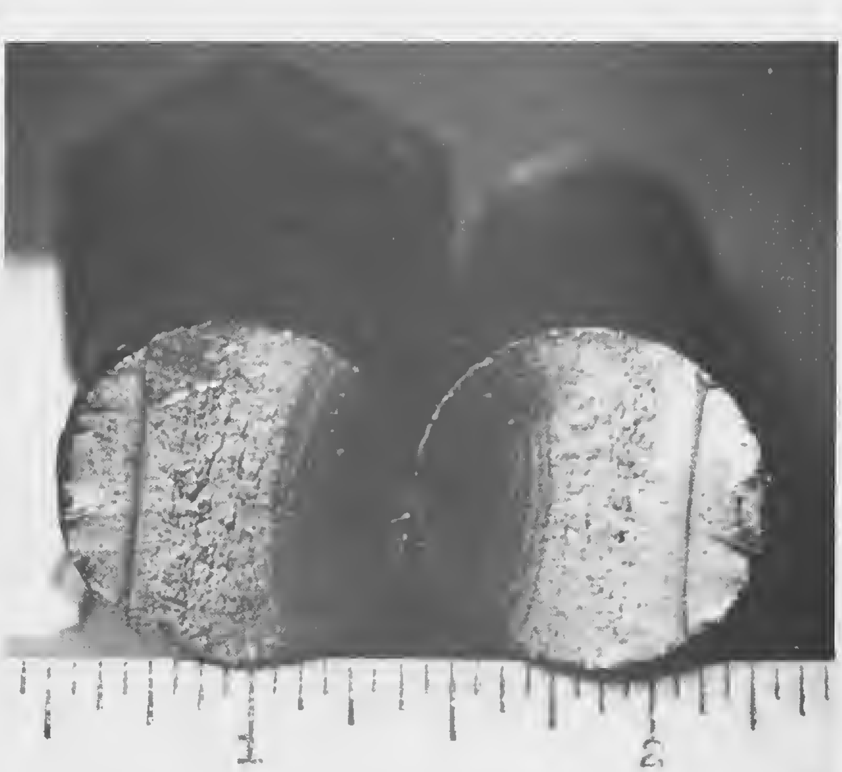


Exhibit 9: Fractured Grade 5 Pivot Bolt.



<u>Exhibit</u> <u>10</u>: Fractured Grade 5 Pivot Bolt.

Lance O'Leary graduated from the University of California at Berkeley in 1953 with a B.S. degree in Mechanical Engineering. Including his periods of employment while still in school, he had worked for the International Harvester Company*, then located in Emeryville, California, for three and one-half years doing truck chassis design work before joining Dynasham in 1955. In March 1958 he had been in the development group associated with suspension design at Dynasham for two years. In this time he had developed two off-highway suspensions and one on/off-highway suspension. He had worked on the development of the Dynasham air suspension (which is for highway use) since its introduction in the middle of 1957. Lance said that the bolt and bushing failures on the air suspension are typical of the problems he encounters while developing a suspension.

The Dynasham air suspension is described in Exhibit 1. It has been available since August 1957 as an option at the rear driving axles of Dynasham's tractor trucks. The air suspension weighs 50 to 70 lbs. more than a conventional steel-spring suspension and also costs considerably more, but it has important advantages, particularly for trucks which must travel one-way unloaded. An unloaded truck and trailer may weigh only a tenth as much as when loaded. If the usual steel leaf springs are used, they must be stiff enough so that their deflection is not excessive when the truck is fully loaded. They are then so stiff that they provide little cushioning when running without cargo. Not only is this uncomfortable for the drivers, but modern lightweight trucks may be susceptible to damage from jarring or road shock when unloaded. Lance mentioned that this is especially true of frameless aluminum tank trucks, which are quite fragile.

An air suspension gives a variable spring rate. In the Dynasham air suspension, the steel springs are replaced by rubber air bags; high pressure air is supplied to the bags by the compressed air system already present on all trucks of this type for actuating the brakes. Dynasham's system is quite simple; height control valves which sense the axle position to the chassis frame control the inlet or release of air. These keep the truck at constant height regardless of load. A time delay is incorporated so that only axle movements due to load changes result in the addition or release of air.

Part of an assembly drawing for the single axle air suspension is shown in Exhibit 2. The suspension was originally designed by Empyrean Supply Company of Milwaukee for non-driving trailer axles. The patterns for the castings were made by Empyrean Supply but the foundry work is done for Dynasham by a local supplier. The two springs per axle, item 24 in the drawing, act as trailing arms (Empyrean Supply terms them radius rods) to locate the axle. They are the only locating links, and

*Real name.

DYNASHAM TRUCK COMPANY, LTD. (B)

Truck Suspension Bolt and Bushing Failures

Bushing Failures

The first of the changes Lance O'Leary specified in an effort to increase bushing life was that longer capscrews be used in the tapped holes in the adjusting blocks. These would fill the holes completely so that the rubber could not extrude into them and tear.

Because Lance did not want to go to the expense of revising the adjustment block and spring hanger castings to remove the sharp corners on their bores, he decided to see if a machined chamfer on the corners of the bores would suffice to protect the bushings. He planned a series of tests, also trying different rubber compounds. Lance pointed out that natural rubber is much harder than neoprene and resists tearing better, thus it was an obvious choice for a better bushing. Intending to specify a larger natural rubber bushing that would be under compression when properly installed, he combined accelerated wear tests on various compounds recommended to him by a vendor with tests of the effect of chamfering the bore corners. The wear tests, primarily for tear resistance, were performed on an assembly consisting of a spring hanger, an adjustment block with bushings and bolt, and the lower spring leaf. Lance said, "I clamped the spring hanger upside down to a bench and pumped the spring leaf up and down by hand, using it as a lever to rotate the adjustment block through its maximum arc of travel. I did this until my arm gave out, which usually took about half an hour, counting strokes and cycling bushings made from rubber of each compound identically."

Lance said that while he would have very much liked to use a mechanical device for testing the rubber bushings, the pressure of time and available manpower ruled out a more elaborate test program. He pointed out that improvisational skills are helpful to an engineer who must operate without the support of extensive back-up facilities.

When his test program was completed, he specified a bushing molded from natural rubber, an ASTM R830G compound.* He explained that molded rubber was found superior to extruded rubber in both tensile strength and tear resistance. He also specified a 1/8 x 45° chamfer on the bore corners of the adjustment block and spring hanger. Machining time costs about \$6 per hour at Dynasham. Although his tests indicated a chamfer was inferior to a radiused corner, Lance decided the chamfer would suffice. A drawing of the molded natural rubber bushing appears in Exhibit 1. At first its dimensions were 3-3/4 inches long by 1-1/2 inches in outside diameter (uncompressed). In May its size was changed; the O.D. was increased to 1-9/16 inches and the length decreased to that of the original neoprene bushing, 3-1/2 inches, to decrease the volume of rubber and reduce mushrooming of the bushing outside the bore of the hanger legs.

While performing the tests, Lance noted that the pivot bolt did not turn when the spring was oscillated provided that the washers were pulled tightly against the hanger legs. If, however, the washers were not in contact with the casting, the bolt turned with each spring oscillation through an angle about half that of the spring movement.

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* The ASTM designation is interpreted as follows:

-- Prefix Letter: "Type R - Compounds made from natural rubber, reclaimed rubber, synthetic rubber, or rubber-like materials, alone or in combination, for services where specific resistance to the action of petroleum-base fluids is not required."

-- 3 Digit Grade Numbers: The first digit indicates the Durometer hardness range, in this instance 80 $\pm$ 5. Durometer hardness is an indentation measure. The second and third digits indicate the minimum tensile strength of the rubber, here 3,000 psi. The tensile strength of the original extruded neoprene bushing was 1,500 psi.

-- Suffix Letter: The "G" signifies a requirement for tear resistance, to be decided upon by the purchaser and supplier, there being no specified ASTM test for this. The bushing material Lance chose had a tear resistance of 500 lb/in, while the original neoprene bushing had a tear resistance of 150 lb/in.

(Source: <u>ASTM Standards</u> - 1961, Part II, Pages 307,308.)

By March 1959, although the extent of the improvement in bushing life had not been fully determined because of the usual difficulty in getting data from the field, Lance thought it obvious that further increases in life were needed, particularly as the applications of the air suspension were being extended to trucks of greater load capacity for use on interstate highways. Some of these trucks would weigh (in total, with trailers) as much as 127,000 lbs., an increase of as much as 50,000 lbs. Axle ratings would remain unchanged, but Lance said that loads on the pivot bolt and bushing would still be greater than before, particularly braking loads. Typical wear of the molded natural rubber bushings is shown in Exhibit 2. Mileages of these bushings are unknown, but well under 100,000 miles.

Of the failures of new bushings that Lance knew of in March 1959, many had been traced to improper installation -- either the pivot bolts had not been fully tightened and the rubber was not under sufficient compression or else shorter than specified capscrews had been used in the two tapped holes in the adjustment blocks.

By March 1959, 308 tandem air suspensions (with dual driving axles) were in service, together with 67 singles and 5 tag units. A tag unit is a dual axle suspension in which only one of the axles drives.

<u>Pivot Bolt Failures</u>

In August 1958, the pivot bolt material specification was changed from SAE Grade 5 to SAE Grade 8. A drawing of the bolt appears in Exhibit 3. The Grade 8 bolts have the same carbon content as the Grade 5 bolts but they are a fine grained alloy steel, quenched in oil and tempered at 800°F minimum. Minimum yield strength is 124,000 psi; minimum tensile strength, 150,000 psi, and the hardness specification is Rockwell C 32 to 38.

At the time of the bolt change, Dynasham mounted a "campaign" through their distributors to get truckers to have the Grade 5 bolts in their air suspension units replaced with the new Grade 8 bolts. Sixty-three Grade 5 bolts were replaced during the campaign, which was instigated when Lance concluded that many suspensions had been built with bolts which might not even have met the Grade 5 specification.

Three failures of Grade 8 bolts had been reported to Dynasham by March 1959. Lance said that all three bolts appeared to have been on the low side of the Grade 8 hardness tolerance band. He thought that the fatigue crack on one of the bolts had originated at a surface defect. All three failures occurred under conditions of particular severity -- two on trucks hauling wood chips and the other on a truck with an end dump semi-trailer used on construction work. Lance explained that these two applications were typical of the rough usage to which the air suspension would be subjected in the future.

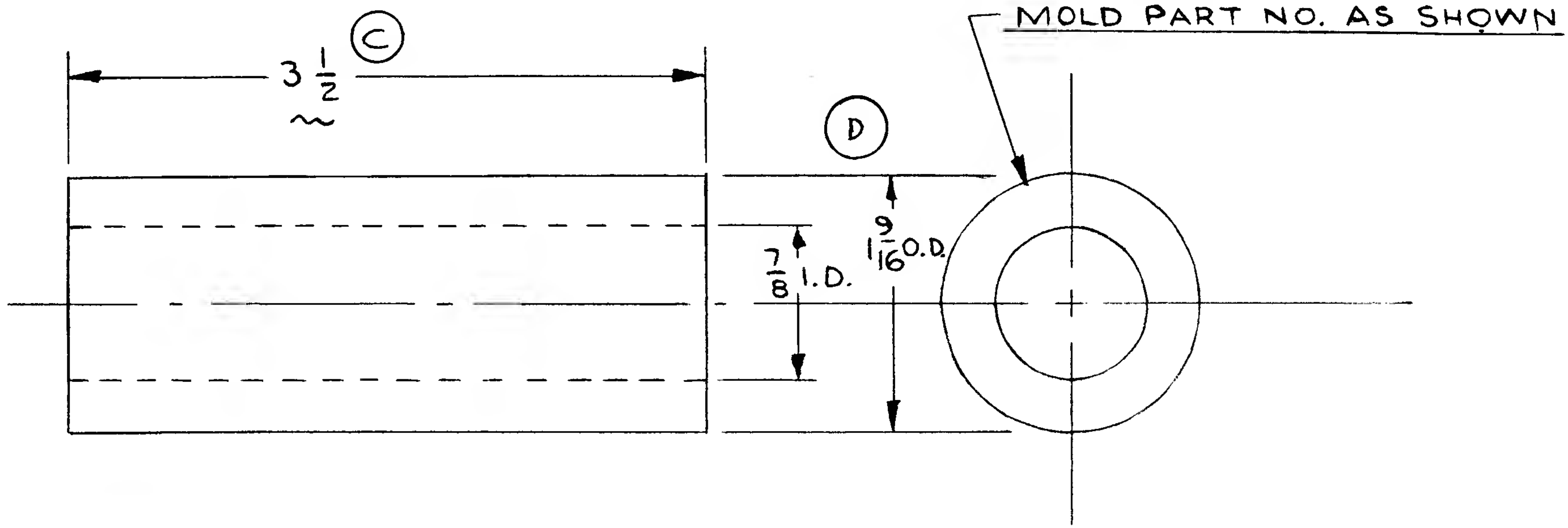
Photographs of a broken bolt from one of the chip trucks appear in Exhibit 4. Lance said that chip trucks have an unusually high center of gravity which results in higher than normal suspension loads. He noted that the harder bolts appeared to show sharper and cleaner breaks than the Grade 5 bolts. Photographs of the broken bolt from the truck with end-dump trailer appear in Exhibit 5. Lance has had several batches of Grade 8 bolts checked for hardness -- one batch averaged about RC 31 to 32, another RC 41 to 42. While Lance did not know the mileages of the three broken Grade 8 bolts at failure, he felt that improvement in bolt life was needed, especially in view of the future demands to be placed on the air suspension.

The list of air suspension failures in Exhibit 6 reveals further the sorts of things Lance had been concerned with during development, as does the report shown in Exhibit 7. This report was written by Lance to Dynasham's Chief Engineer. It is not typical of Lance's job, being the first written report by him concerning the air suspension, but was occasioned by the unusual persistance of complaints by the owner of the truck.

Originally 1-1/16 inch (inside diameter) standard SAE washers (.136 inch to .192 inch thick) were used on the 7/8 inch pivot bolt. Lance said that this gave a sloppy fit and that the washers also were bent to a dished shape by the force exerted upon them by the compressed rubber. In March 1959, Empyrean Supply began making a special washer for Dynasham to eliminate this problem. These washers are 1/4 inch thick, with an inside diameter of 15/16 inch and an outside diameter of 2-1/2 inches. Lance said, "These shouldn't bend and will help keep the rubber compressed."

Lance said that at this time he felt one of the underlying reasons for the bolt failures to be insufficient rubber to adequately absorb shock loads. He also thought that the bolt diameter was too small; he felt it was unfortunate that, keeping the same castings with their fixed bore diameters, an improvement in one of these areas would be cancelled, or worse, in the other. He said, "I thought perhaps a greater volume of active rubber in twist was needed in the bushing, or that perhaps the bushing shouldn't be twisted. However, if the rubber is restricted to the bore of the adjusting block, with no rubber in the hanger legs, the bolt would be subjected to high shear forces where it left the hanger legs. It seemed to me that the rubber in the legs protected the bolts from failure in shear."

He began debating whether to recommend design changes which would entail new spring hanger and/or adjusting block castings.



MATERIAL
ASTM R830G
KIRKHILL COMPOUND # 180C1519

SUPERSEDES DWG.

Exhibit 1: Bushing.

LET.	DATE	BY	REVISIONS	REFERENCE
D		BUCK	WAS 1½ O.D.	
C		BUCK	3½ WAS 3¾	
			NEOP.	
			WAS 60-70 SHORE	
B		FRED	REDRAWN & MAT	

ITEM	PART NO.	NAME	NO. REQ.

THIS DRAWING AND INFORMATION HEREON IS THE PROPERTY OF

MOLDED RUBBER BUSHING FOR FRONT SPRING HANGER CONN.

TOL. UNLESS SPEC DEC ± .001 FRAC 1/32 ANG ± 1/2°

SCALE	MAT.	DATE
FULL	NOTED	

PART NUMBER	DWN BY	CHKD	DWG NO	REV
	FRED			D

ECL 44-B
Exhibit 2

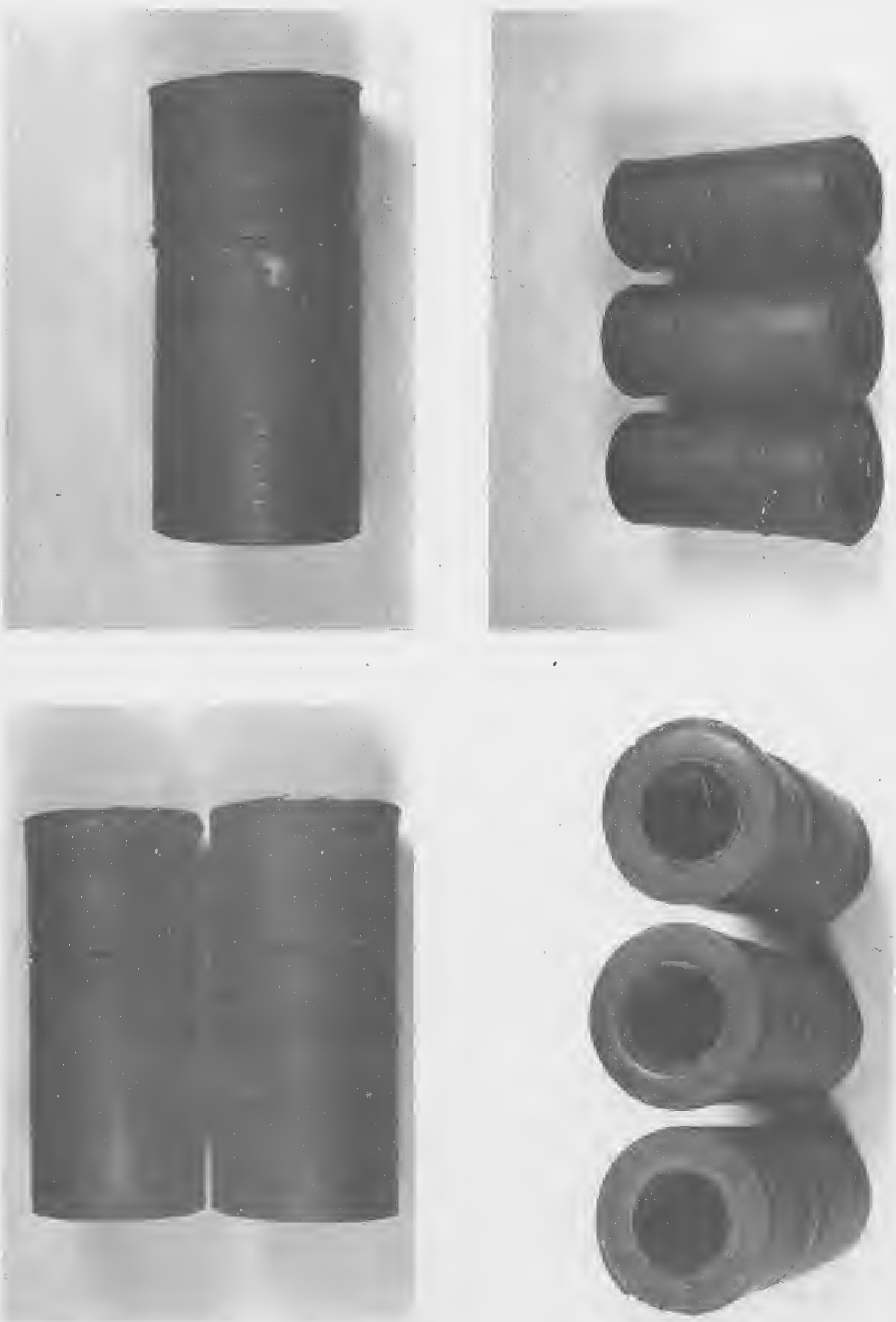
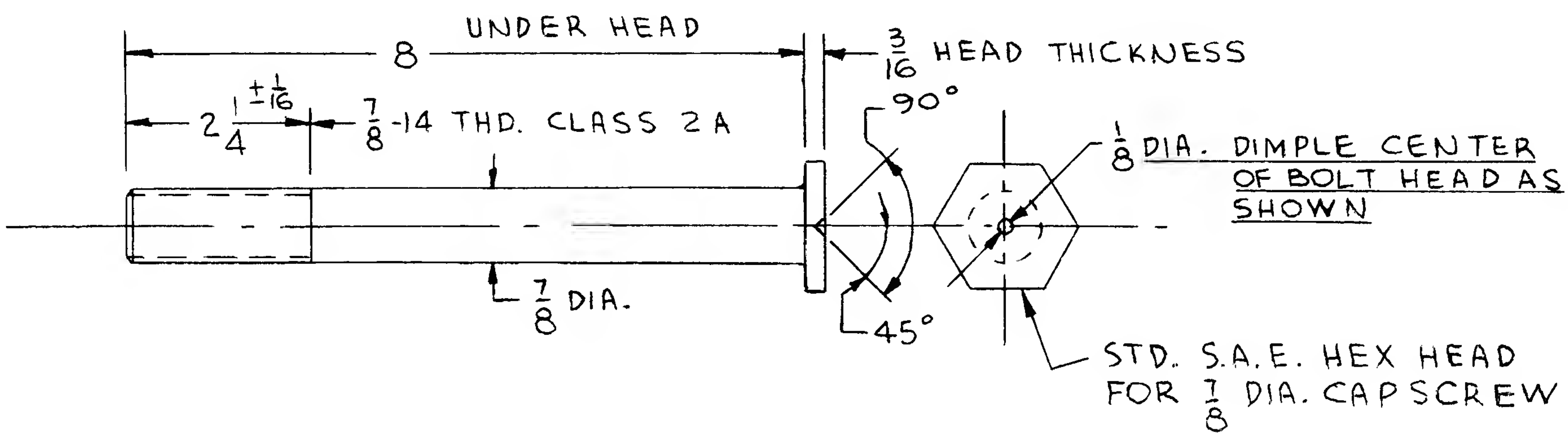


Exhibit 2: Worn Molded Natural Rubber Bushings. 3 1/2 inches Long.



CAPSCREW MATERIAL
4140 CLASS 8

Exhibit 3: Pivot Bolt.

ITEM	PART NO.	NAME	NO. REQ.

THIS DRAWING AND INFORMATION HEREON IS THE PROPERTY OF

CAPSCREW

TOL. UNLESS SPEC	DEG. ± .015	FRAC ± 1/32	ANG ± 1/2°
SCALE 1/2	MAT. NOTED	DATE	
PART NUMBER	DWN BY	CHKD.	DWG NO.

REV.

LET	DATE	BY	REVISIONS	REFERENCE
D		BUCK	REV. "C" OMITTED;	
C		BUCK	NOTE ADDED;	
			CLASS 5;	
B		BUCK	REDRAWN; WAS MAT.	

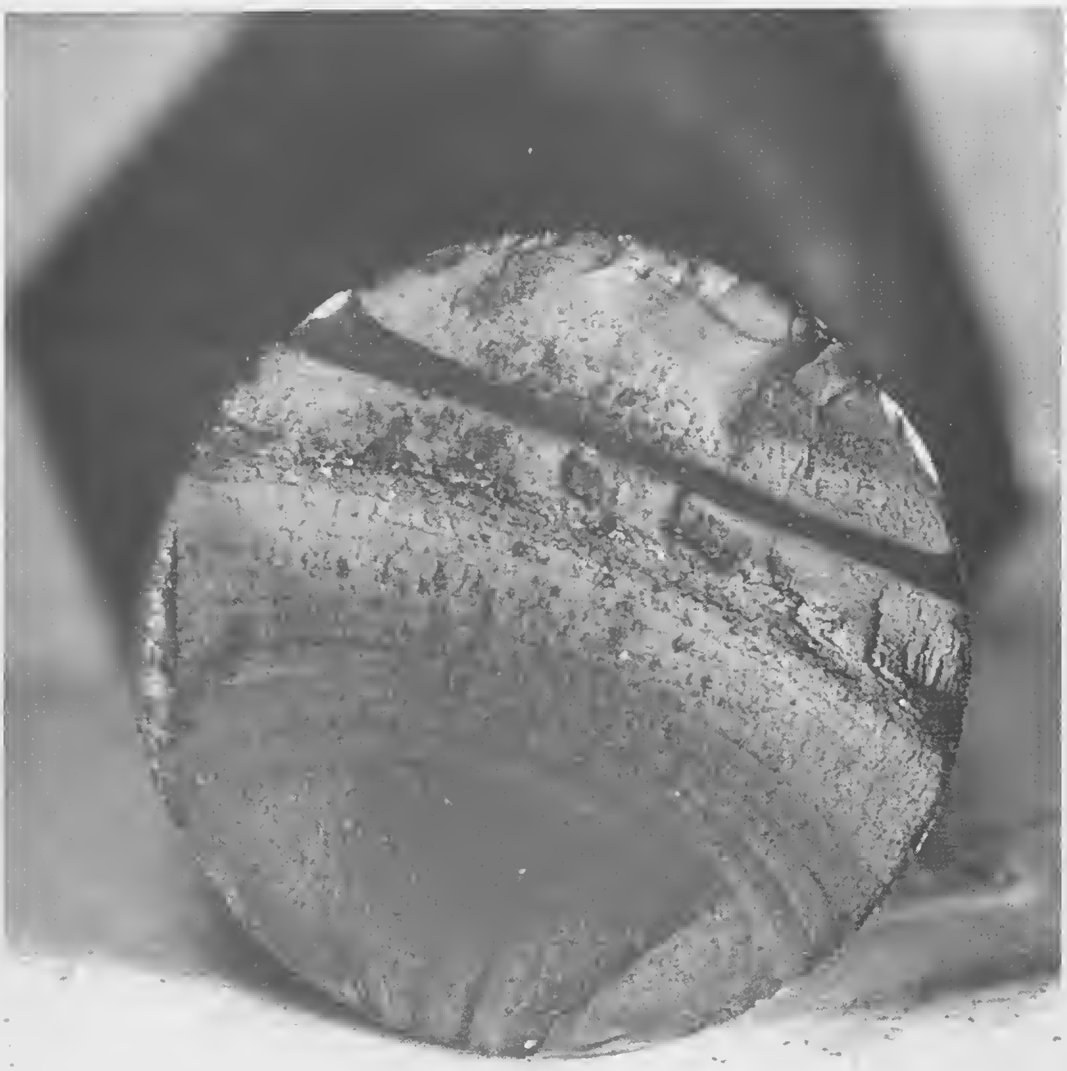


Exhibit 4: Fractured Grade 8 Pivot Bolt from Chip Truck.

Exhibit 5: Fractured Grade 8 Pivot Bolt from End Dump Semi-Truck. Bolt Bent by Lance O'Leary to Check Ductility.

Exhibit 6: Failures Involving the Dynasham Air Suspension

March 23, 1959

As of February 1959, 380 trucks have been delivered with Dynasham's air suspension. 308 of these units are equipped with tandem drives, while the remainder 72 are single drives. Of these 380 trucks, approximately 100 have had failures. Below is a list of the total parts failures of these units.

Attached you will find further information on these failures.

Bushings	23	
Pressure Plates	20	
Leveling Valves	12	
U-bolts	15	*F-7 **R-10
Rough Riders	9	
Air Springs	4	
Spring Leaf	5	
Reported out of adj.	8	
Reported leaking air	5	
Spring Drive Bolts	32	
Spring Drive Bolt Campaign	63	
Suspension Alignment	2	
New Bolt Failures	3	
Cracked Frame Rails	5	
TOTAL	206	
CAMPAIGN	- 63	
TOTAL PARTS FAILURES	143	

Enclosure: 1

*F - Front
**R - Rear

Exhibit 7: Report Written by Lance O'Leary to Dynasham's Chief Engineer

To: Charles Skallman

From: Lance O'Leary March 30, 1959

INSPECTION OF AIR SUSPENSIONS REBUILT BY REPAIR SHOP
Truck: X.L. 3977418 (Stock - Auto Truck)
Owner: Sam Spade
Mileage: 148,000

The owner has repeatedly complained about the suspension giving a rough ride and having poor handling characteristics. He says the suspension is unstable both in tracking and in roll. He has complained about those conditions since he bought the truck. The Service Department has the history of the complaints and the action taken to solve the problem.

When the truck was brought in last week, I was asked by Service to analyze the problem and advise corrective action. The current complaint was rough ride, unstable handling characteristics and the rear drive axle off-tracking the forward unit. The rear unit appeared to be out of track by approximately 3/4". The shock absorber bushings were badly cut by this offset condition.

I wasn't able to tell what was causing the axle to off-track. All of the components appeared to be sound with the exception of the shock absorber mounting and this was the result of the offset axle. I suggested to Mr. Fox that we could update the suspension by revising the shock absorber mountings and at the time the suspension was dismantled, I would be able to analyze each part to determine the cause of the problem. The decision was made to strip the suspension from the frame and check each piece.

The following comments are my observations from my inspection of each part of the suspension:

1. The spring seats were located and welded as prescribed.
2. The springs were OK except the center bolts had been loosened. I had new bolts installed.
3. The pivot adjustment blocks on the front drive axle were to print but the two on the rear drive were not. The machined surface was not parallel to the bore. This condition would cause the axle to off-track.
4. The spring hanger brackets for the front drive axle were to print but the pair for the rear were not. The portion of the spring hanger bracket that locates the spring in relation to the frame was not adequate to hold the cores in the proper location. This apparently happened on these which further aggravated the axle off-track.

5. One shock absorber had lost its fluid and was ineffective. This would have a decided effect upon the ride quality, as well as the roll stability.
6. The cap screws that hold the adjustment block to the spring are to be of sufficient length to fill the tapped holes in the block. All the cap screws were 1/4" short and allowed the rubber bushings to work into the tapped holes and therefore lose their effective compression. This would account for:

    a. Unstable steering control
    b. False track setting
    c. Short bushing life

7. Three aluminum and one steel U-bolt saddles were on the suspension with all of the aluminum saddles broken beneath the rear U-bolt. The U-bolt nuts were frozen and both conditions would indicate that the U-bolts were not properly tightened and the axles were allowed to work in the suspension and therefore shorten the life of the rubber bushings, as well as the pivot bolts.

The suspension was reassembled with new parts replacing those found defective. The suspension was readjusted and all the problems appear to be solved. The owner is to report on the handling and riding characteristics to determine the effect of these last efforts.

DYNASHAM TRUCK COMPANY, LTD. (C)

Truck Suspension Bolt and Bushing Failures

In October 1959, Dynasham decided upon a second set of modifications to the pivot bolt connection. The new design used a one inch Grade 8 bolt, up from 7/8 inch. No changes were made to the spring hanger and adjustment block castings. Two natural rubber bushings were used with unchanged compounding, but they were made shorter, with a greater I.D. and separated by a nylon spacer as shown in Exhibit 1. The new bushing is shown in Exhibit 2 and the spacer in Exhibit 3. The washers are unchanged except for an I.D. increased to 1-1/16 inch.

The Ru-Glyde used during assembly is a soap-like lubricant. The torque specification for the pivot bolt (100 ft-lbs) was determined during a test program carried out by Empyrean Supply Company. A pivot bolt tightened to this figure compresses the rubber bushings so that a torque of about 100 ft-lbs must be exerted on the adjustment block before the bushings will slip as the block twists. Lance O'Leary and Empyrean Supply adopted as an indicator of bushing condition the torque required to twist the adjustment block. They decided to set 60 ft-lbs as the lower limit for a good bushing, while 20 ft-lbs or less was taken as indicating failure. The interval from 20 to 60 ft-lbs was left as a doubtful zone.

The standard life test developed at Empyrean Supply combined twisting of the adjustment block relative to the spring hanger with reversing loads on the block. The spring hanger was held and a load of 8,000 lbs applied to the adjustment block; then the block was rotated through 15° and a load of 4,000 lbs in the opposite direction applied. Raynal Penrose, Empyrean Supply's Chief Engineer explained, "During braking the coefficient of friction may reach one -- then each spring would see a load of 8,000 lbs. We thought the reverse loading of 4,000 lbs at 15° rotation would be an adequate representation of the drive torque reaction."

The first tests were to an assembly of the original design with neoprene bushings, as described in Part A. Failure occurred at 29,000 cycles, as determined by disassembly and inspection. Next, the design of Part B, with natural rubber bushings and a 7/8 inch Grade 8 bolt was tested; by 44,000 cycles the torque had fallen below 20 ft-lbs.

Lance said, "We had been trying to compress two long bushings by pressing on their ends, and I don't think we were getting much compression in the middle. Then, during the first few cycles of motion the bushing would work in the castings and the amount of compression would tend to equalize. Because the compression was so localized to begin with, I think we were ending up with pretty loose bushings."

Raynal Penrose explained the thinking behind the new design: "We felt that if we could compress the rubber equally throughout, then it would carry the load equally. So next we cut the rubber shorter, to 2-1/2 inches, and put a one inch thick polyethylene washer on each side of the bushings, so we had: washer, natural rubber, natural rubber, washer. This ran for 73,000 cycles before the bolt fractured from fatigue. We were happy to see that the rubber was still in good shape. Since the bolt broke we concluded that the test was close to field experience."

Next Empyrean Supply checked the stress in a 7/8 inch bolt, finding the maximum to be about 102,000 psi. At this stress they calculated that a Grade 8 bolt should fail in reverse bending at 70,000 to 100,000 cycles. Raynal said, "This seemed to prove our analysis correct. The next logical step was to turn the assembly inside out; we tried a two inch long steel spacer between two natural rubber bushings using a one inch bolt. The spacer was made of steel to save on cost. This should have given us a maximum stress of 60,000 psi with infinite fatigue life, but at 38,000 cycles the bolt broke. We found that the edge of the steel spacer had worn a stress raiser into the bolt, so then we changed the spacer material to nylon.

Assemblies of the new design, with nylon spacers, gave torque resistances of 60 to 80 ft-lbs after 150,000 cycles and 30 ft-lbs after 250,000 cycles. No further fractures of one inch bolts occurred. Lance said, "Not only does the spacer decrease the length of rubber we have to compress, it also stabilizes the bolt and limits its stress and deflection in bending. With this design the torque drops rapidly from 100 ft-lbs during the first 1,000 cycles or so, but then seems to level off at 60 to 80."

Raynal said, "It's interesting that as we run more cycles on the natural rubber bushings, they seem to work harden; this has led us to believe we are approaching an asymptotic limit and that longer tests would show no further deterioration."

In September the spacer material specification had been changed from nylon to nylatron because nylatron is less expensive and works as well.

Lance considered several alternatives before deciding to use the bushing-spacer-bushing design. All the others would have required new castings or machining operations, or both, as well as more expensive purchased parts. One alternative was the design shown in Exhibit 4, which Lance drew up for several heavy duty, special order trucks. Within the adjustment block is a 1-1/8 inch bolt and a 3-piece bushing with rubber bonded to the inner sleeve and contained within the outer sleeve. The bushing is made by the Lord Manufacturing Company* and when Lance ordered twenty, their price was $20 apiece. He found that even in the largest quantities Dynasham might order, the cost would still be $9 each. The spring hanger legs are machined to accept compressed delrin bushings and hardened wear-washers are used between the Lord bushing and the delrin bushings. This design required new adjustment block castings and the machining on the hanger legs had to be held to close tolerances to get the desired compression of the delrin. Four trucks went into service with these installations in August 1959.

Since this design was obviously too expensive for general adoption, Lance drew up the similar but lower-cost alternative shown in Exhibit 5. A Clevite-Harris*Silentbloc bushing which costs only $3 would be used. With a one inch pivot bolt, the original adjustment block castings could be retained, but their bores would have to be machined to take the bushing. Delrin bushings are again shown in the hanger legs, but Lance used a stepped washer instead of a counterbore to cut costs. Lance also drew up a design using a Clevite bushing with a 1-1/8 inch bolt that would interchange completely with the Lord bushing design; for this installation new castings would be required. No assemblies with Clevite bushings were built.

Lance said, "When we saw that the one inch bolt and spacer worked out okay, the choice was obvious, since our cost is the same as with the old design and the parts are all interchangeable with earlier air suspensions."

*Real name.

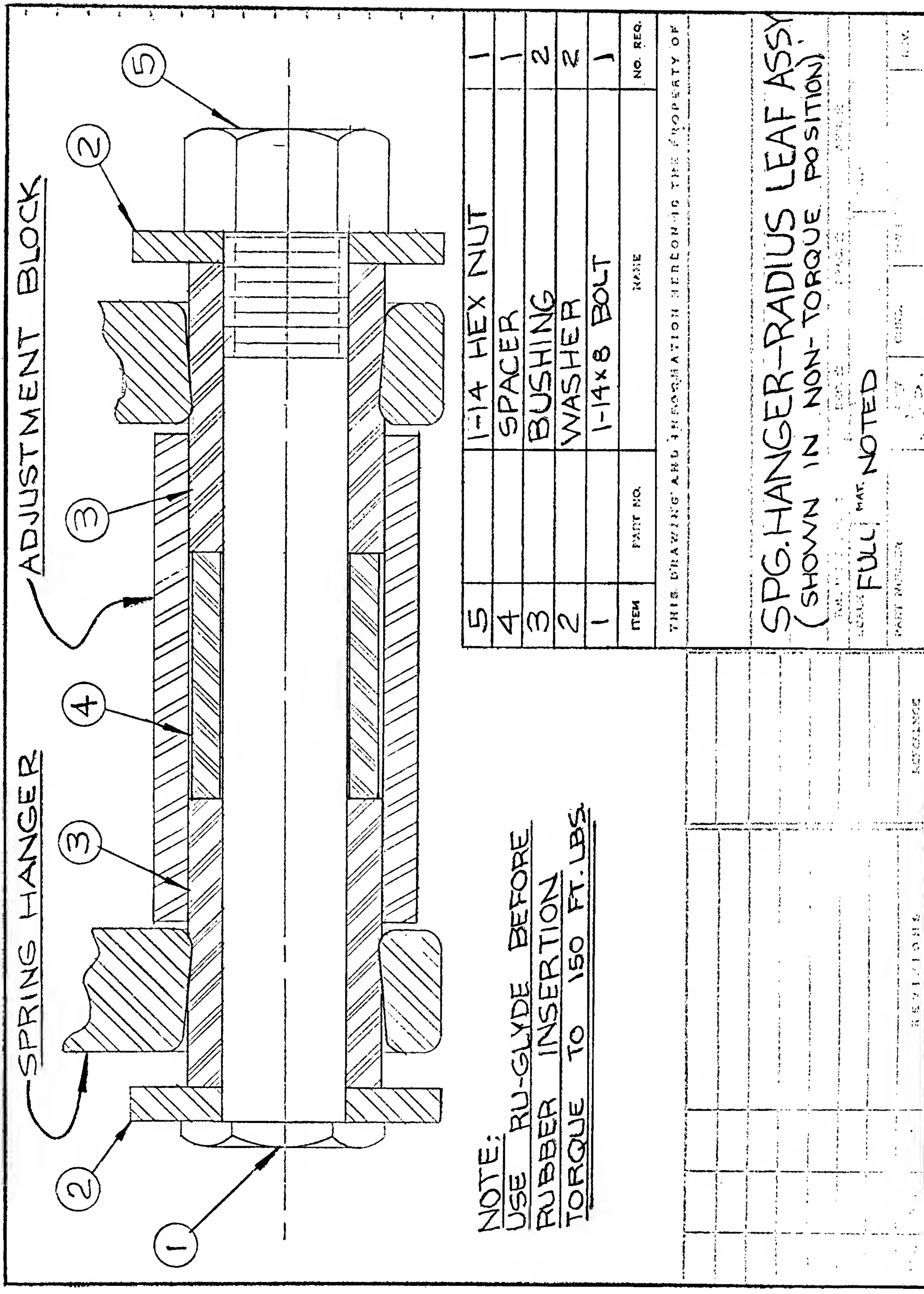


Exhibit 1: Bushing-Spacer-Bushing Assembly.

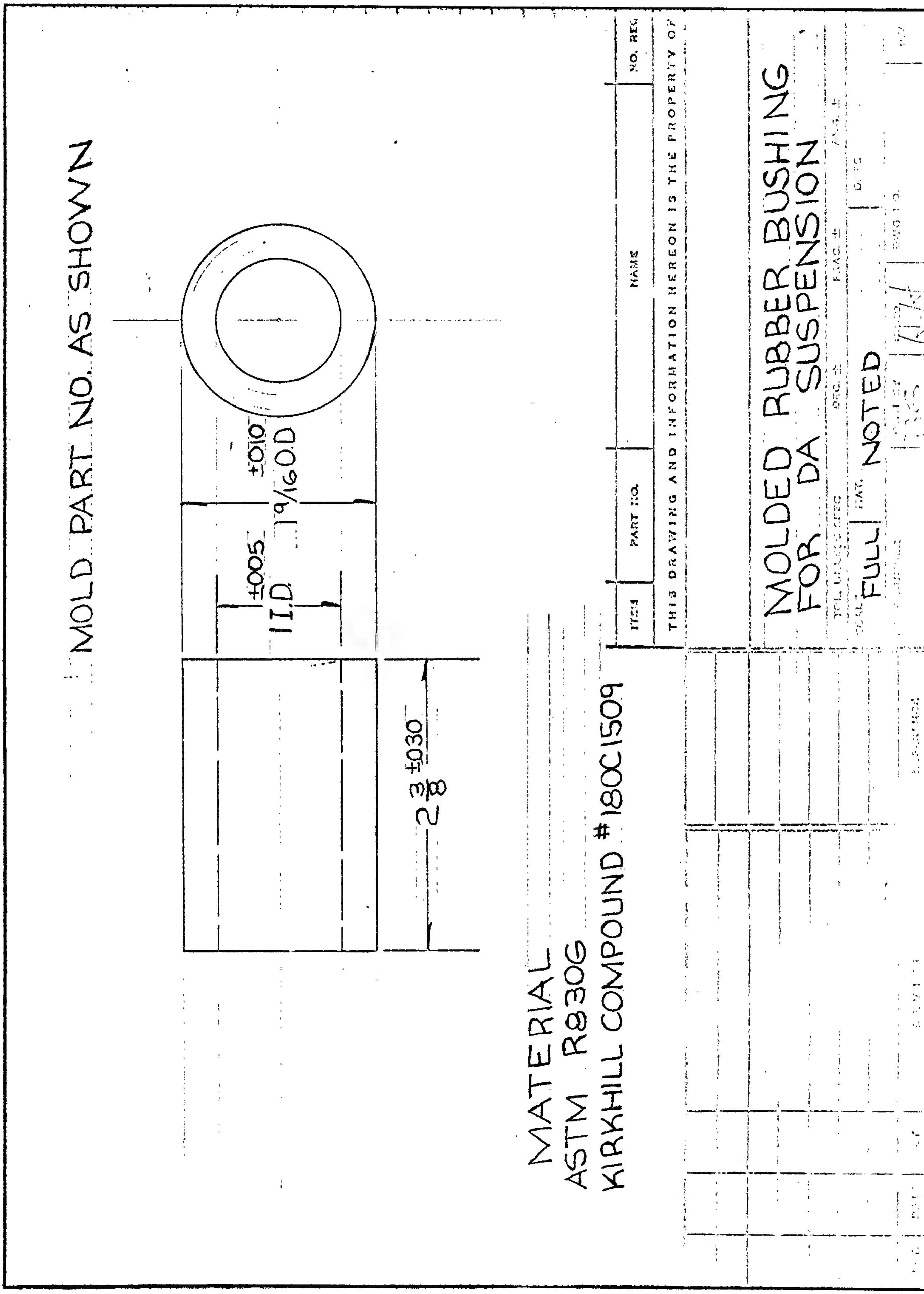


Exhibit 2: Bushing.

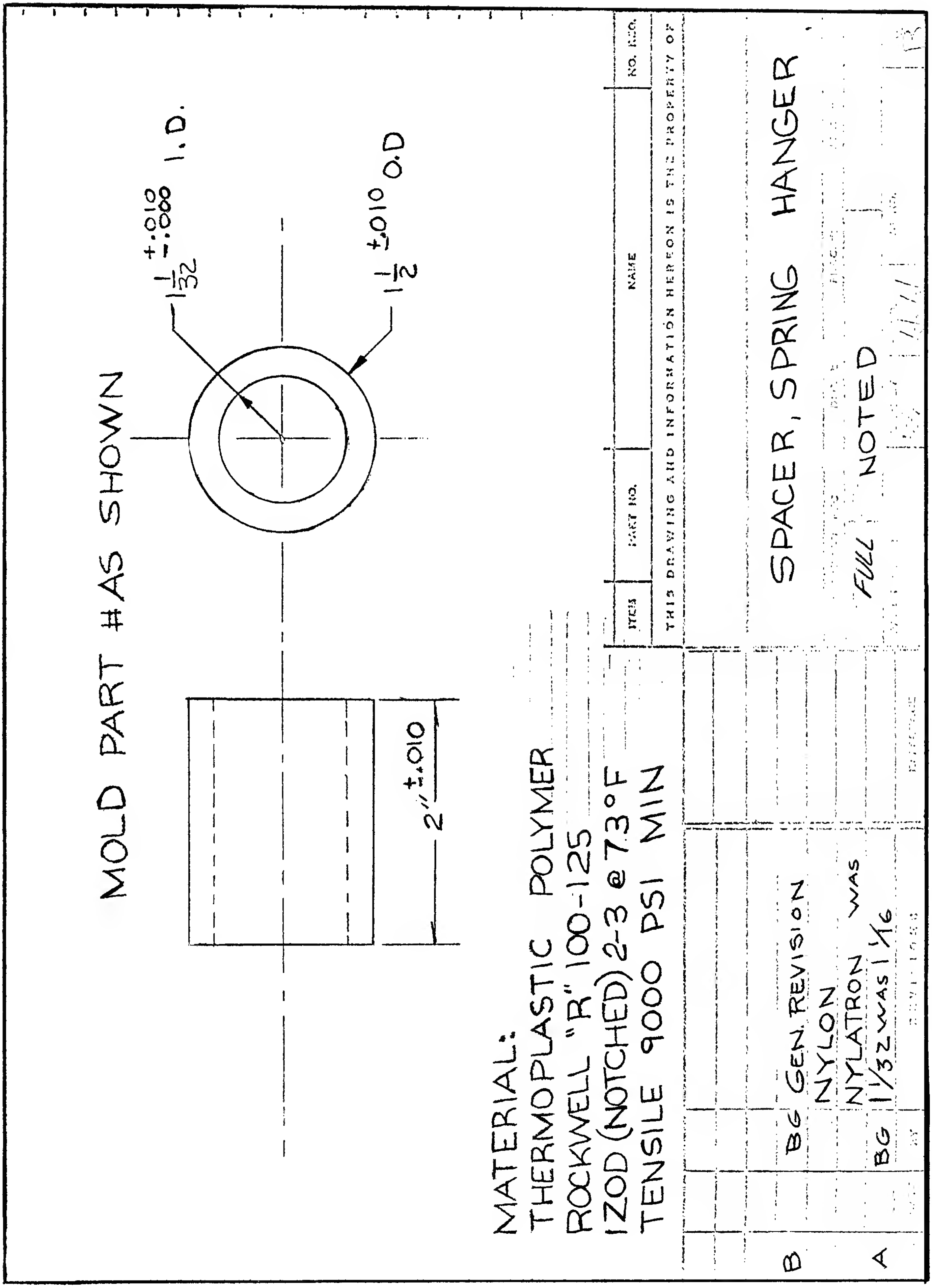


Exhibit 3: Nylatron Spacer.

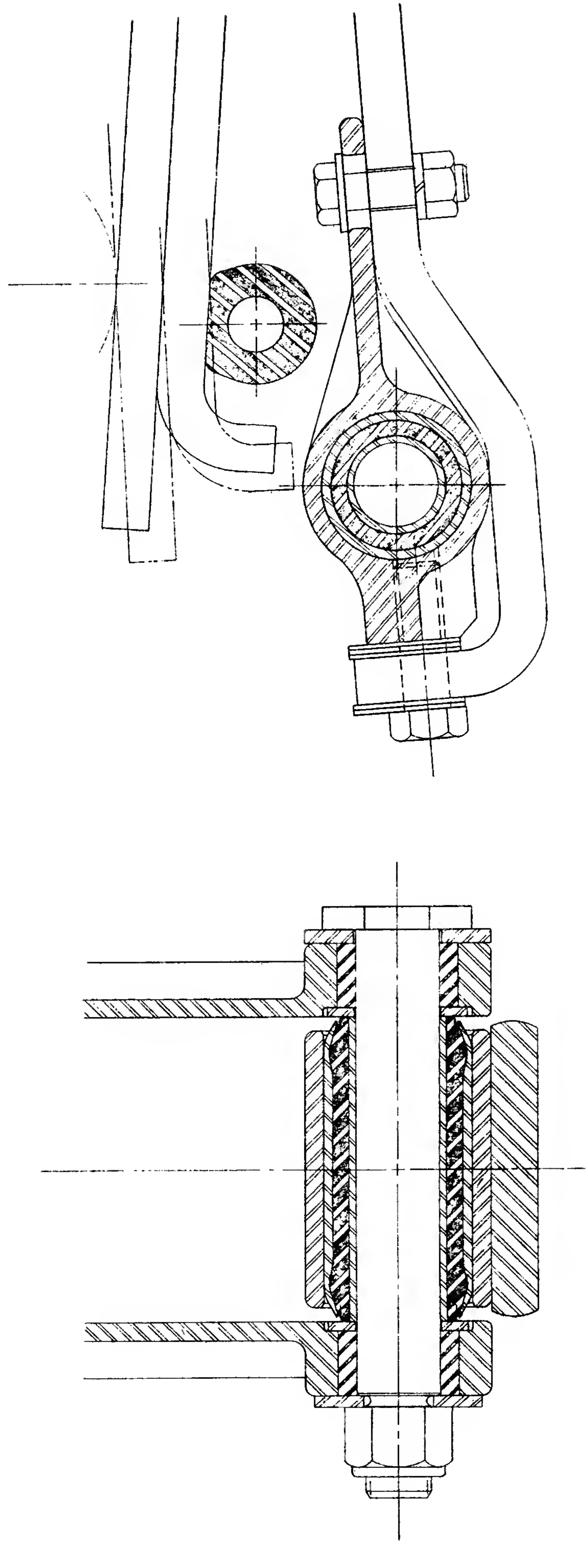
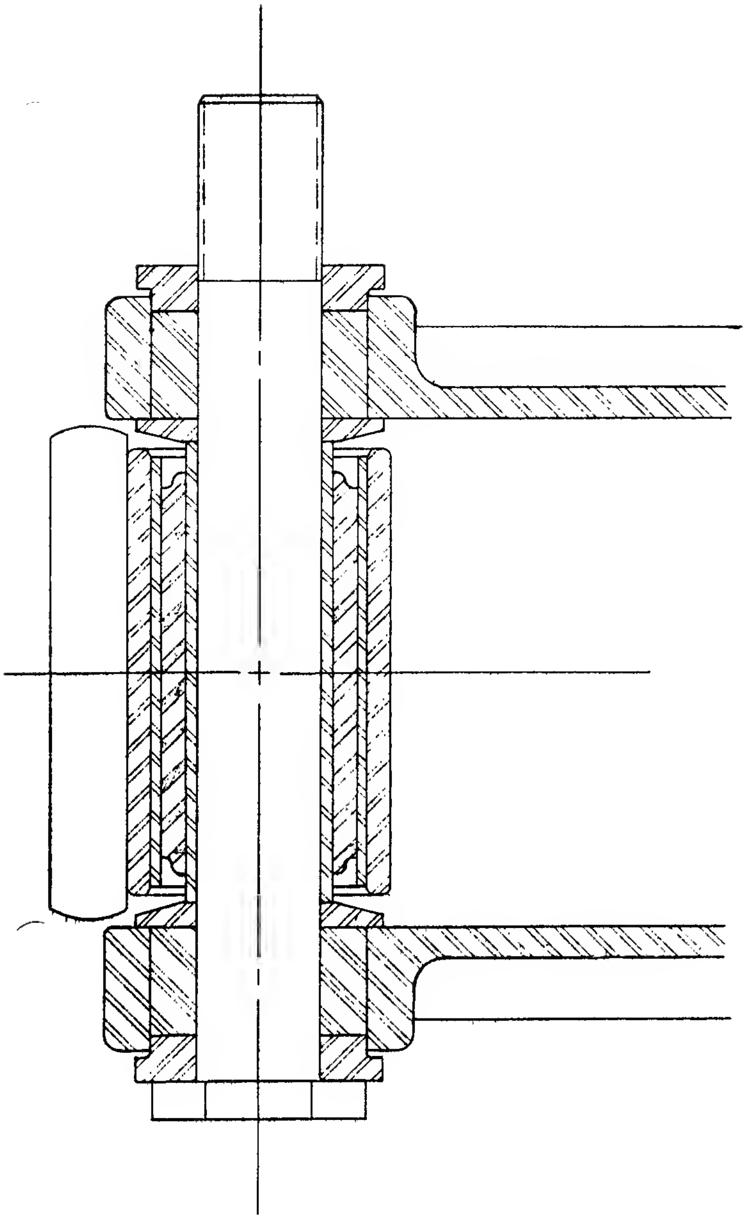


Exhibit 4: Lord Bushing Assembly.



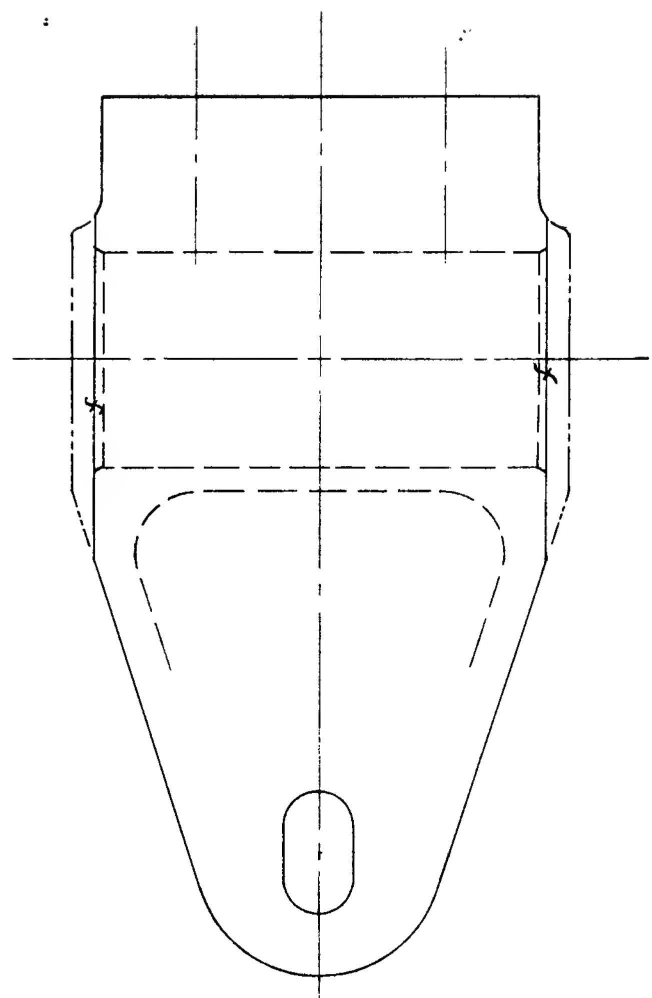


Exhibit 5: Clevite-Harris Bushing Assembly.